Active control of compressor noise in the machinery room of refrigerators

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The low-frequency noise generated by the vibration of the compressor in the machinery room of refrigerators is considered as annoying sound. Active noise control is used to reduce this noise without any change in the design of the compressor in the machinery room. In configuring the control system, various signals are measured and analyzed to select the reference signal that best represents the compressor noise. As the space inside the machinery room is small, the size of a speaker is limited, and the magnitude of the controller transfer function is designed to be small at low frequencies, the controller uses FIR filter structure converged by the FxLMS algorithm using the pre-measured time signal. To manage the convergence speed for each frequency, the frequency-weighting function is applied to FxLMS algorithm. A series of measurements is performed to design the controller and to evaluate the control performance. After the control, the sound power transmitted by the refrigerator is reduced by 9 dB at the first dominant frequency (408 Hz in this case) and 3 dB at the second dominant frequency (459 Hz here), and the overall sound power decreases by 2.6 dB. Through this study, an active control system for the noise generated by refrigerator compressors is established. © 2019 Institute of Noise Control Engineering.

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1 INTRODUCTION

As people's perception of living standards and noise increases, interest in sound comfort and reduced noise from home life is increasing. Many studies to reduce the noise generated from household appliances have been conducted\textsuperscript{1,2}. In particular, since the refrigerator operates all day, researchers are trying to reduce various noise generated by the refrigerator. Kim et al.\textsuperscript{3,4} reduced the noise generated by the refrigerator by changing the structure of the refrigerant pipe and created a noise map of the refrigerant to predict flow noise, and Heo et al.\textsuperscript{5,6} researched the blade pass frequency (BPF) noise of the centrifugal fan of the refrigerator and changed the design to reduce the noise.

However, among noises generated by a refrigerator, the low-frequency compressor noise is frequently described as an unpleasant noise\textsuperscript{7}, and the noise reduction cannot be effectively achieved by passive method, e.g., sound absorption and/or damping. Therefore, studies were conducted to reduce the noise by changing the structure of the machinery room, the shape of the compressor, and the mounts\textsuperscript{8–10}. Kim et al.\textsuperscript{11} reduced the noise by modifying the stiffness and mass of the shell of a reciprocating compressor, and Lee et al.\textsuperscript{12} modified the shape of the joint of the discharge case of the linear compressor to reduce the noise.

In addition to ineffective reduction performance of passive methods at low frequencies, the passive methods redesigning the compressor and/or the machinery room require high costs. Without the change of the current design, the compressor noise can be controlled using active noise control effectively at low frequencies. Many researchers implemented noise attenuation using active control for various applications\textsuperscript{13–19}, but it is hardly found that active control is applied for refrigerator application.

In similar studies, active noise control was applied to enclosure structures\textsuperscript{20,21}. Lee et al.\textsuperscript{22} attempted to control the noise propagated through the opening of enclosure structures surrounding the sound source using feedforward control. Ji et al.\textsuperscript{23} also tried to control the noise propagated from the enclosure structure using causal finite impulse response (FIR) filter with an additional time delay according to the harmonics fundamental frequency. In two studies on enclosure structures, the entire transfer function between the disturbance and the reference signal within the frequency range of interest can be measured. Also, the loudspeaker size is not limited to satisfy the space of the enclosure, and the controller can be designed regardless of the target frequency. Nevertheless, in the case of a refrigerator machinery room, a compressor, pipe, and blower fan
are disposed therein. Thus, space for a control loudspeaker has to be small, and a method of measuring transfer functions between the reference and disturbance signals is not easy. Under this constrained condition, Koo et al. measured the transfer function when the refrigerant and the machinery room fan were not operating. However, in the case of a real refrigerator, the transfer function of the noise source cannot be secured by the method as in Ref. because the transfer function changes with time.

In this study, a reference signal and a loudspeaker are selected for the practical control system, and the controller is designed to reduce the acoustic power radiated from the machinery room. A time domain controller design method is utilized, and the design method of the controller is studied when the size of the loudspeaker is small because of the limited space inside the machinery room. To design the controller, the transfer function of the loudspeaker is measured, and an FIR filter is designed using the sound pressure and acceleration time signal. Experiments are carried out by installing the designed controller on the digital signal processor (DSP) equipment, and the control performance is verified.

2 CONTROL SYSTEM CONFIGURATION

2.1 Selection of the Reference Signal

Figure 1 is a schematic of an active control system for the compressor noise generated in a refrigerator machinery room. In the compressor located on the left side of the machine room, noise is generated owing to vibrations, and the noise is transmitted through the left window and the right window. The light gray squares inside the machinery room represent the window of each side. The compressor operates at an operating frequency of 51 Hz, and the spectrum of the sound pressure when operating at that frequency is shown in Fig. 2. The sound pressure generated by the compressor is large at 408 Hz and 459 Hz. Therefore, the control target frequencies are selected as 408 Hz and 459 Hz.

To reduce this noise component, a feedforward control system is chosen; the control system consists of a reference signal sensor, an actuator, an error sensor, and a controller. In this section, reference signal sensors, actuators, and error sensor selection are described. The controller design is discussed in the next section.

To find an appropriate reference signal, we measured various signals related to the refrigerator noise. As candidates for the reference signals related to the noise, the driving voltage signal to the compressor, signals measured by an accelerometer attached to the compressor shell, and signals of the accelerometer attached to the machine window were selected.

The transfer function and the coherence of the disturbance for the three reference signals were calculated, and time-dependent changes of the transfer function at control frequencies of 408 Hz and 459 Hz are shown in Fig. 3 and Fig. 4, respectively. When the voltage used to drive the compressor is the reference signal, it can be seen in Fig. 3 and Fig. 4 that the fluctuation width of the magnitude and phase of the transfer function is large and the coherence is low. If the accelerometer signal attached to the machine room window is the reference signal, it can be seen in Fig. 3 and Fig. 4 that the transfer function also fluctuates and the coherence is not enough for the control. When the accelerometer signal attached to the compressor shell is the reference signal, the variation of the transfer function with time is almost zero, and the coherence is always higher than 0.95. Therefore, the accelerometer attached to the compressor shell was selected as the reference signal sensor to control the noise generated by the compressor.

Figure 5 shows the components of the machine room to be controlled. A loudspeaker is installed on the left front part near the window, and a small loudspeaker is selected because of the internal space problem. A reliable accelerometer in the frequency band of interest was selected with calibration data. The coherence between the acceleration signal and the sound pressure outside the window was compared with each location, and the accelerometer was attached to the upper right part of the compressor shell with the highest coherence.

2.2 Selection of the Actuator

In order to perform active noise control, it is important that the sound pressure generated by the speakers is linearly proportional to the applied voltage to achieve the control performance. However, as large voltage is applied to the loudspeaker relatively to their size, the transfer function of the loudspeaker can be nonlinear. Therefore, the loudspeaker performance should be considered for designing the controller to exhibit the control performance.

There was not enough space in the machinery room, it cannot be possible to choose loudspeaker with appropriately designed cabinet. The global control performance would be improved with a monopole-like secondary loudspeaker having the cabinet. However, if we use the loudspeaker
having the cabinet which should be smaller loudspeaker to fit the space, it would be difficult to satisfy that the driver works in the 400 Hz band, which is the main target frequency of this control. Thus, in this article, the 3-inch loudspeaker shown in Fig. 6(a) was selected. The frequency response of the loudspeaker with white noise excitation is shown in Fig. 6(b). Since the resonant frequency of the control source is 260 Hz, this device is not designed to work below this frequency. Peaks below 260 Hz are noise amplification. This can be confirmed by the coherence of Fig. 6(c).

Generally, the frequency response of the loudspeaker is measured using swept sine method. Because this method ensures a separation of linear and non-linear part of the sound generation, this allows a more accurate measurement of the frequency response function. Nevertheless, there are two reasons for measuring the frequency response using white noise in this article. First, this procedure was considered for mass-produced refrigerators. White noise was considered appropriate to reduce production time since system identification would be required for tuning on the production line. Second, as shown in Fig. 6(c), we can obtain frequency response with high coherence in the control band even with white noise. This is because white noise with low RMS value is applied to the loudspeaker, resulting in a small non-linear part.

![Figure 2](image2.png)

**Fig. 2**—Sound pressure spectra of the compressor noise.

![Figure 3](image3.png)

**Fig. 3**—Time history of the value of transfer function and coherence at 408 Hz from the reference signals to the error sensors: input voltage (blue solid), compressor shell acceleration (green dashed), and machinery room window acceleration (black dotted).
2.3 Selection of the Error Sensor

A microphone is used as an error sensor for evaluating the active control performance by the controller. Microphones are selected as instruments capable of measuring the frequency range of interest according to calibration data. The position of the error sensor is selected as the center point of the left and right windows of the machine room window. There are slits on the left, slits on the right, and other small holes in the noise path of the machine room. However, except for the left slit and the right slit, others are excluded from the controller design because the cross-section of those holes is small and the contribution to the noise transmission is low. The error sensor is placed at the center of both windows because the transfer function change according to the error sensor position in the window of the machine room is not significant within the frequency of interest.

3 CONTROLLER DESIGN THEORY

3.1 FIR Filter Design Using the FxLMS Algorithm

In general, the filtered-x least mean squared (FxLMS) algorithm converges to a controller by minimizing an error signal using a reference signal $x(k)$ and an error signal $e(k)$ measured in real time. However, in this article, the controller used in the experiment was designed as off-line using the signal measured in advance. That is, the FxLMS algorithm was only used to design the controller, and the use of microphones is eliminated in actual experiments. Thus, the actual control experiments were conducted in a simple feedforward system without a feedback loop. The controller is converged by the FxLMS algorithm by using an acceleration signal and a sound pressure signal, which are measured in advance.

The block diagram of the FxLMS algorithm is shown in Fig. 7. The reference signal $x(k)$ generates a disturbance $d(k)$ through the plant $H_P(z)$, and the control source $y(k)$ is generated through the controller $C(z)$ updated with an LMS filter and the secondary path transfer function $H_s(z)$ of the control loudspeaker. The control source and the disturbance act in opposite phases. The LMS filter receives the filtered reference signal $x'(k)$ and the error signal $e(k)$, which is the difference between the disturbance and the control signal at the error microphone. Then, filter converges the controller $C(z)$ in a direction to minimize the

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**Fig. 4**—Time history of the value of transfer function and coherence at 459 Hz from the reference signals to the error sensors: input voltage (blue solid), compressor shell acceleration (green dashed), and machinery room window acceleration (black dotted).

**Fig. 5**—Component inside machinery room.
The equation for updating the coefficient of the controller can be expressed as Eqn. (1).

\[ C(k+1) = C(k) - \mu \frac{\partial J}{\partial C(k)} \quad (1) \]

Here, \( k \) is sample index, and \( \mu \) is the convergence coefficient of the controller, where the cost function \( J = e(k)^2 \) and the error signal \( e(k) \) is expressed as:

\[ e(k) = d(k) - y(k) \ast H_s(k) \quad (2) \]

where \( \ast \) denotes linear convolution, \( y(k) = [y(k) y(k-1) \ldots y(k-L+1)] \), \( y(k) \) is filter output and \( L \) is the filter length when \( H_s(z) \) is expressed in the form of an FIR filter.

When Eqn. (2) is substituted into Eqn. (1), the controller coefficients of the next discrete time converge as shown in Eqn. (3).

\[ C(k+1) = C(k) + 2\mu x'(k)e(k) \quad (3) \]

The larger the value of \( \mu \), the faster the filter coefficient converges, but the filter coefficient diverges when \( \mu \) is larger than a certain value. Therefore, the value of \( \mu \) should be appropriately selected within the following range\(^2^7\):

\[ 0 < \mu < \frac{2}{||S||^2 \sigma_s^2 (L + 2D_{eq})} \quad (4) \]

where \( \sigma_s^2 \) is the reference signal power, \( ||S|| \) denotes the norm of the secondary path FIR coefficients, \( L \) is the filter length and \( D_{eq} \) is equivalent delay parameter of the secondary path.

### 3.2 Frequency-weighted FxLMS Algorithm for Controller Convergence Rate Control

Generally, the FxLMS algorithm can be used to design an FIR filter that minimizes the acoustic power measured by an error sensor. However, when using the FxLMS algorithm, it converges from the frequency which have high power of the disturbance signal and the reference signal. If the response of the converged FIR filter is large at a low frequency and the loudspeaker performance at a low frequency is not sufficient for generating a control source, harmonic distortion may occur at the control frequency. In order to avoid this situation, an appropriate frequency-weighting filter is applied to the reference signal and the disturbance signal to slow down a convergence rate of the controller at a low frequency and to increase the convergence speed of the controller at the target frequency.

Figure 8 shows a block diagram of the FxLMS algorithm with a frequency-weighting filter. \( W(z) \) is the FIR filter coefficient of the frequency-weighting function. Since the reference signal and the disturbance signal through \( W(z) \) have a high magnitude in the target frequency band, the control coefficients converge faster in the target band. In addition, since the weighting function is applied to both signals, noise can be controlled without distortion of the

![Fig. 6](image)

*Fig. 6—Experimental setup (a), frequency response (b) and the coherence (c) of the sound pressure to the voltage applied to the loudspeaker.*

![Fig. 7](image)

*Fig. 7—Block diagram of FxLMS algorithm.*
Moreover, by using the frequency-weighting function, it is possible to increase the convergence speed of the controller at a frequency where the magnitude of the both signals are small.

Numerical simulation is conducted to verify the effect of the frequency-weighting function. The reference signal and the disturbance signal are harmonic signals having a fundamental frequency of 150 Hz. The magnitudes of each frequencies of the reference and disturbance signals for convergence through the FxLMS algorithm are shown in Table 1. To set a different convergence speed at each frequency, the magnitude difference between 150 Hz and the other two frequency are set to be 10 times or more.

After convergence of the controller, the magnitude of the response of the controller without applying the frequency-weighting function is shown in Fig. 9(a). Also, the magnitude of the reference signal is shown in Fig. 9(b), and the sound pressure before and after using this controller is shown in Fig. 9(c).

In the case of the controller converged with the FxLMS algorithm without the weighting filter, the controller converges only at 150 Hz, which is large in both signals. Even at 300 Hz, the controller slightly converges, showing some control performance when compared to that before and after the control. However, in the case of the 450 Hz component, the controller hardly converges. In Fig. 9(b), the magnitude of the control signal at 150 Hz is too large. Therefore, when a small loudspeaker is used in an actual control experiment, the control performance may be adversely affected by the non-linear part of the low frequencies.

To overcome this phenomenon, the converged controller with the weighting filter of Fig. 10 has to be used. The frequency-weighting function consists of non-zero values at 300 Hz and 450 Hz in the frequency domain. The magnitude of each sinusoids are 0.5. The frequency weighting was implemented as an FIR filter, and FIR filter was designed by the frequency sampling method. The same procedure is repeated with the frequency-weighting function. The applied result is shown in Fig. 11. As a result of weighting function, the filtered reference signal and the

### Table 1—Reference signal and pressure magnitude of 150 Hz harmonics to verify effect of frequency weighting function.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>150 Hz</th>
<th>300 Hz</th>
<th>450 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference magnitude (V)</td>
<td>0.1 V</td>
<td>0.01 V</td>
<td>0.001 V</td>
</tr>
<tr>
<td>Pressure magnitude (Pa)</td>
<td>0.01 Pa</td>
<td>0.001 Pa</td>
<td>0.01 Pa</td>
</tr>
</tbody>
</table>

![Fig. 8—Block diagram of frequency weighted FxLMS algorithm.](image)

![Fig. 9—Before applying frequency weighting function: (a) Magnitude of the controller, (b) Magnitude of the reference signal, and (c) Sound pressure spectra.](image)
disturbance signal are reduced at frequencies other than the two frequencies.

The control performance is high at target frequencies of 300 Hz and 450 Hz, and the amplitude of the control signal at 150 Hz is small. Therefore, it is possible to control the convergence speed of the controller for each frequency band by applying an appropriate frequency-weighting function. By this method, it is prevented that a high voltage at a low frequency is applied to the small loudspeaker through the method of converging the controller above 250 Hz.

4 EXPERIMENT OF SOUND POWER CONTROL

4.1 Experiment Equipment and Configuration

Figure 12 shows the experimental setup for the controller design and the acoustic power measurement before and after the control. The internal configuration of the machinery room is shown in Fig. 5. The reference signal and the error signal were measured in an anechoic chamber to design the active controller. The position of the microphone is shown in Fig. 12(a). It is installed at the center of the left window and the right window outside the machine room cover. A charge-type accelerometer is used to measure the acceleration value as the reference signal for designing the controller. B&K Nexus conditioning amplifier was used to connect the acceleration signal measured by the accelerometer to the dSpace RTI 1103 in the form of voltage. To reduce aliasing occurring in the DSP, the acceleration signal was passed through a low-pass filter with a cut-off frequency of 1000 Hz. Low-pass filter SR640 of the Standard Research System was used, and LMS SCADAS III was used to measure the reference signal and the sound pressure. The schematic diagram of this procedure is shown in Fig. 13. The sampling time of measured time signal and applied signal to DSP was set to 1/5120 s, and this

![Graph](image.png)

**Fig. 10**—Frequency weighting function to adjust convergence rate at 150 Hz, 300 Hz, and 450 Hz.

**Fig. 11**—After applying frequency weighting function: (a) Magnitude of the controller, (b) Magnitude of the reference signal, and (c) Sound pressure spectra.
time step has a reduction of more than 30 dB at the frequency at which aliasing would occur.

The performance of the designed controller was evaluated in both anechoic chamber and reverberation chamber. First, the sound pressure before and after the control was measured on both windows of the machinery room and on evaluation point of the refrigerator. After that, to measure the acoustic power control performance of the designed controller, the refrigerator was moved to the reverberation chamber. The change in the acoustic power before and after the control was measured by the reverberation method. The installed refrigerator and microphone are shown in Fig. 12 (b).

### 4.2 Frequency Response Function

#### Measurement and Controller Design

The frequency response function (FRF) of loudspeaker $H_s(z)$ is the sound pressure at the error sensor position relative to the voltage signal applied to the loudspeaker. The sound pressure and voltage were measured to calculate the FRF. The measurement point of sound pressure is the same as the microphone position in Fig. 12(a). Using the function generator, white noise with 1 V of peak value was generated and applied to the speaker. During the measurement, both the fan and the compressor in the refrigerator were stopped. The frequency response function was calculated by dividing the sound pressure of microphone by voltage applied to the loudspeaker. Averaging method was used to reduce measurement noise. The number of averages is 100, and the overlap percentage is 99%. We set the sampling frequency to be 5120 Hz and the frequency resolution to be 2.5 Hz. The measurement results of the FRF are shown in Fig. 14. The magnitudes of the loudspeaker FRF $H_s(z)$ are sufficient at above 250 Hz as in Sec. 2.2, and it is confirmed that the phase is not accurately measured owing to low performance at frequencies below 250 Hz.

Because of this characteristic, the controller transfer function of the corresponding frequency should be set to nearly 0, so that the control sound pressure is not generated below 250 Hz. Also, the magnitude of controller transfer function should not be large at a frequency of 250 Hz or greater.

In order to control the noise of the compressor, the controller was designed by measuring the time signal using the equipment described in Sec. 4.1. The reference signal and the disturbance signal were measured during the operation of the refrigerator. The spectrum of the measured signal is shown in Fig. 15 along with the spectrum of the signal using the frequency-weighting filter. The weighting function consists of frequencies of 408, 459, and 500–800 Hz, where the frequencies are harmonic components of 51 Hz. The weighting function is applied in the form of FIR filter, and the filter length is 2048. Also, it is designed by frequency sampling method.

![Experimental setup for designing controller (a) and for measuring sound power difference before/after control (b).](image)

**Fig. 12**—Experimental setup for designing controller (a) and for measuring sound power difference before/after control (b).

![Schematic diagram to measuring acceleration signal for using reference signal.](image)

**Fig. 13**—Schematic diagram to measuring acceleration signal for using reference signal.
Time-domain and frequency-domain response of the weighting function are shown in Fig. 16. It can be seen that both signals are large only at the frequency to which the weighting function is applied.

The controller is converged using the FxLMS algorithm, which is described in Sec. 3, for time signals with and without the application of the weighting filter. The length of the FIR filter to be converged is set to 2048. The frequency response of the converged controller is shown in Fig. 17. In the case without weighting function, all of the harmonic components of the operating frequency of 51 Hz have magnitude values. However, the response of the controller using the weighted time signal has amplitudes of 408 Hz and 459 Hz, which are the control target frequencies, and it is confirmed that the magnitude also appears between 600 Hz and 800 Hz, which is the target of the weighting function. By applying the weighting function, the amplitude of the controller can be made small at below 250 Hz, so it can prevent a situation where a high voltage is applied to the loudspeaker at a low frequency.

4.3 Control Experiment and Result

The control performance of the refrigerator was measured when the controller was turned on/off, and the measurement was conducted in the anechoic room and the reverberation room. The instrument setup during the measurement is shown in Fig. 12.

Fig. 14—Magnitude and phase of the transfer function to the control speaker at left window (solid) and right window (dashed).

Fig. 17—Spectra of reference signal (a) and disturbance signal (b) with/without frequency weighting function.
Figure 18 shows the experimental results of the control using the designed controller described in Sec. 4.2. The sound pressure in the left window near the noise source compressor is reduced by about 9 dB at 408 Hz and by 17 dB at 459 Hz. The sound pressure measured in the right window decreased by 6 dB at 408 Hz and increased by 3 dB at 459 Hz. The peaks of the mid-frequency noise are generated by the machine room fan. Because this fan is closer to the right window, the mid-frequency noise measured in the right window is louder than left window. Since this noise is relatively smaller than the compressor noise, there is little influence on the change in the overall noise after the control.

In addition, in general, the noise of the refrigerator is evaluated at a distance of 1 m from the back of the refrigerator. The change in the sound pressure before and after the control is plotted in Fig. 19. At the rear at 1 m, the maximum sound pressure is at 459 Hz, and the sound pressure decreases by 5 dB after the control. The control performance of the 408 Hz component is measured at 2 dB.

Finally, the control performance was verified by measuring the change of the sound power before and after the control in the reverberation room. The change in the acoustic power before and after the control is shown in Fig. 20 for the 1/3 octave band. The control frequencies of 400 Hz
and 500 Hz are reduced by 9 dB and 3 dB, respectively, and the overall acoustic power is reduced by 2.6 dB.

5 CONCLUSION

In this study, a method for designing an active controller for mass-product, limited-space refrigerator is studied. The noise is generated mainly by the compressor in a refrigerator machinery room. The time domain method is used to design the controller. To adjust the convergence speed, the time signal is filtered using the frequency-weighting function. The designed controller is implemented into DSP equipment. A series of experiments is conducted to verify the performance. The following conclusions are then obtained:

1) Acceleration of the compressor shell is the most suitable reference signal for control system of refrigerator. Various signals are compared as a reference signal that correlated with the compressor noise. The coherence and the transfer function are considered for the most appropriate reference signal. The coherence between the acceleration and the sound pressure is always high regardless of time, and the transfer function is hardly changed over time. As a result, the corresponding signal is selected as the reference signal.

2) Frequency-weighting filter is suggested for adjusting convergence speed at each frequency of FxLMS algorithm when using small loudspeaker. The space inside the refrigerator is narrow, only small loudspeakers are available for the control. As a result, the control frequency is limited to 200 Hz or greater. To implement these constraints in the controller design, the control target frequency is limited by applying a frequency-weighting filter to the reference signal and the disturbance signal. Simulation and experiment are conducted for verifying performance, and the result of these can prove effectiveness of frequency-weighting function.

3) The control performance can be confirmed through the experiment in an anechoic room and a reverberation room. In the case of the control performance measured in the anechoic chamber, the sound pressure at the window near the compressor is greatly reduced. On the far side, the sound pressure decreases

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**Fig. 19**—Control performance at 1 m behind from refrigerator in the 1/3 octave band.
at 408 Hz but slightly increases at 459 Hz. However, the sound pressure measured at the rear 1 m point, which is the method of evaluating the refrigerator noise in the industrial field, decreases at both frequencies. Moreover, the sound power in the reverberation chamber decreases in the 400-Hz and 500-Hz octave bands. Talking about mid-range frequency noise, this band is dominantly influenced by the fan, so there is no control effect. However, the compressor noise has a higher contribution to overall noise. Thus, after control, the sound power is reduced by about 3 dB.

Through this study, a method to reduce the low-frequency noise generated by a refrigerator compressor by using a loudspeaker of a limited size is established. As a method of designing the controller, a method of using a frequency-weighting function filter is presented, and the convergence speed changes according to the presence or absence of a filter and its effect are confirmed. In addition, the acoustic power reduction performance of the machinery room is verified through the control experiment using an actual refrigerator. In the future, we will study the control performance change according to the number of taps of the filter required for refrigerator commercialization and control model establishment for refrigerator control at various operating frequencies.

6 ACKNOWLEDGMENTS

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7 REFERENCES